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DOES MORE TENSION REDUCE VIV?

Prof. J. Kim Vandiver
Massachusetts Institute of Technology
Cambridge, MA, USA

Leixin Ma
Massachusetts Institute of Technology
Cambridge, MA, USA

ABSTRACT

In this paper it is shown that for very long risers in sheared flow there is a surprising outcome—the VIV response amplitude in the power-in region does not depend at all on the amount of damping in the power out region, as long as it is sufficient to prevent waves from reflecting at the boundary and returning to the power-in region. In these cases, the response in the power-in region depends on the wave radiation damping and not on the damping in the power-out regions. Tension plays a major role in the determining the radiation damping and in some cases, but not all, pulling harder will indeed reduce response in the power-in region. Numerical simulations are presented in which a finite element model of a long riser is used to compute the VIV response in a sheared flow for which the power-in region is at one end of the riser. The radiated waves are shown to diminish with distance traveled as expected. When $\zeta_{out}n_{out}$, the product of the number of wavelengths to reach the far termination and the damping ratio in the power-out region is greater than 0.18, it is shown that no significant vibration energy returns to the power-in region and the response in the power-in region is independent of the damping in the power-out region. The numerical simulation is used to illustrate the effect of changing tension on the radiation damping and therefore on the VIV response. The VIV response prediction program SHEAR7 is used to evaluate the effect of increasing tension on a realistic deepwater drilling riser in 3000 m water depth. A 20% increase in tension leads to a 12% reduction in fatigue damage rate.

Keywords: Flexible tensioned riser; Flow induced vibration; Radiation damping

INTRODUCTION

Drill ship operators have long believed that VIV may be reduced if you pull more top tension. In shallow water this approach may work when it moves the response natural frequency away from resonance with the flow-induced vibration frequency which produces maximum response. At low mode number the separation between natural frequencies is great enough that there will exist current velocities at which maximum VIV response will occur. Specific current velocities, which result in maximum response will be separated by regions of low response. These low response regions correspond to reduced velocities which do not favor large VIV. By changing tension, it is sometimes possible to detune the resonance by changing the natural frequency. For such low-mode number risers, detuning the riser reduces the VIV response amplitude by moving the reduced velocity for the resonant mode to a less favorable value. In sheared flows, at low mode numbers the hydrodynamic damping in power-out regions contributes to the total modal damping for the resonant mode and therefore has a significant effect on the vibration amplitude in both the power-in and the power-out regions.

For very long risers operating at similar current speeds, the excited modes are higher in mode number and the natural frequencies are more closely spaced in frequency. There are multiple potential resonant modes with reduced velocities favorable to large amplitude VIV. It is in general not possible to reduce VIV by attempting to detune the resonant mode by changing tension, because the riser shifts its response to another equally problematic mode at a slightly different but more favorable reduced velocity. A typical response prediction of a very long riser in a linearly sheared flow is shown in Figure 1.

This paper explores the effect of changing tension on the response of long, high mode number risers, such as depicted in Figure 1. The analytical approach taken is to require equilibrium between the power flowing into the flexible cylinder to the power flowing out due to various types of damping. The simple spring-mounted rigid cylinder is used as an example to show that requiring an equilibrium between power-in and power-out reveals the same controlling dimensionless damping parameter, as that previously derived by equating lift and damping forces [Vandiver, 2012]. Power flow is easier to estimate for flexible cylinders than spatially dependent lift forces. The analysis of power flow equilibrium is used to derive a dimensionless damping parameter, which governs the response of very long flexible risers. Radiation damping is shown to be the form of damping which regulates response for risers which behave as if of infinite length. It is shown that in such cases radiation damping increases with tension, and therefore response decreases with increasing tension.

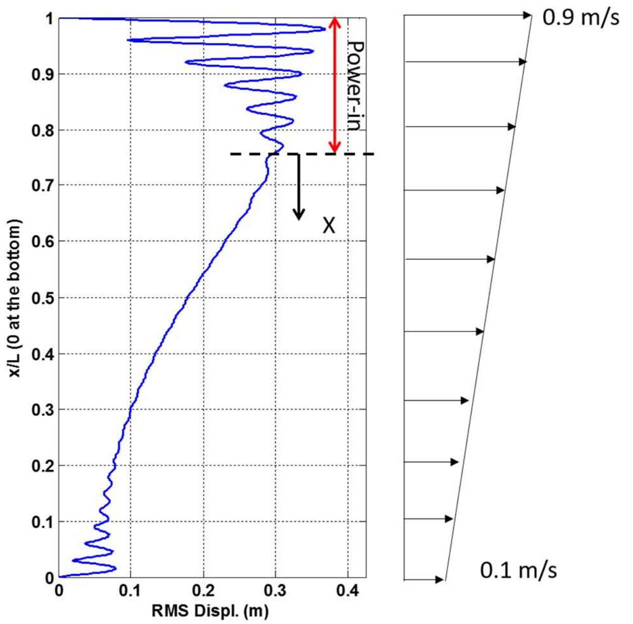

Figure 1. An example of travelling wave response of a long, high mode number riser in a sheared flow

Nomenclature	
$A_{1,rms}$	RMS amplitude for outbound wave
A_{rms}	Spatial and temporal RMS of $y(x,t)$ in the excitation region
A^*	Dimensionless amplitude for rigid cylinders
A^*_{rms}	Dimensionless amplitude for flexible cylinders
$c(x)$	Damping coefficient, damping/length
c_{in}	Damping coefficient in the power-in region
c_{out}	Damping coefficient in the power-out region
c_s	Structural damping coefficient –rigid cylinder
c^*	Dimensionless damping parameter for rigid cylinders or flexible infinite cylinders
C_L	Lift coefficient for rigid cylinders
C_{L0}	The value of the peak lift coefficient
$C_{L,rms}$	RMS lift coefficient in the VIV excitation region
D	Cylinder diameter
$f(x,t)$	Lift force per unit length
f_v	VIV response frequency
k	Stiffness/length
L_{in}	Length of power-in regions
L_{out}	Length of power-out regions
m	Mass/length with added mass
n	Mode number
n_{out}	Number of wavelengths in the excitation region
P	Tension
T	Oscillation period of the pipe
U	Flow speed
U^2_{rms}	Mean square flow velocity in the excitation region
$U_{r,opt}$	Most favorable reduced velocity
x	Axial coordinate along the pipe
$y(x,t)$	VIV crossflow displacement
ζ_{in}	Damping ratio in the power-in region
ζ_{out}	Damping ratio in the power-out region
ρ	Density of fluid
ω	Response frequency in radians / second
$\langle\Pi_{in}\rangle$	Time-averaged input power
$\langle\Pi_{out}\rangle_s$	Time-averaged power flowing out due to structural damping in the power-in region
$\langle\Pi_{out}\rangle_{rad}$	Time-averaged power carried by radiation of waves away from the power-in region
$\langle\Pi_{out}\rangle$	Time-averaged power flowing out
λ	Average wavelength

POWER FLOW FOR A SPRING-MOUNTED CYLINDER

In this paper steady state dynamic equilibrium is formulated in terms of power flow. The simplest VIV system is cross-flow response of a spring-mounted rigid cylinder, which has been the subject of numerous papers over the last six decades. The cylinder is characterized by *m*, c_s, and k, which are the per unit length values of the mass, damping and stiffness. In a typical experiment, a cylinder of total length L and diameter D is exposed to uniform flow with speed U. At steady state the power flowing into the system from lift force excitation must equal the power lost in damping.

The power flowing into the cylinder is expressed in equation 1 and the power-out due to damping is given in equation 2. At steady state the time-averaged power-in from the fluid lift excitation is in equilibrium with the power-out due to damping. Equations 1 and 2 show these power calculations for an harmonic lift force of the form $F(t) = \frac{1}{2}\rho U^2 D C_L L \sin(\omega t)$ and response velocity of the form $\dot{y}(t) = A\omega\sin(\omega t)$.

$$\langle\Pi_{in}\rangle = \frac{1}{T}\int_0^T F(t)\dot{y}(t)dt$$

$$= \frac{1}{T}\int_0^T \frac{1}{2}\rho U^2 D C_L L\sin(\omega t) A\omega\sin(\omega t)\, dt \qquad (1)$$

$$= \frac{1}{4}\rho U^2 D C_L L A\omega$$

$$\langle \Pi_{out} \rangle = \frac{1}{T}\int_0^T c_s L \dot{y}^2(t)dt$$
$$= \frac{1}{T}\int_0^T c_s\, L A^2\omega^2 \sin^2(\omega t)dt = c_s L\frac{A^2\omega^2}{2} \quad (2)$$

Equating equations 1 and 2 and solving for A/D, yields a single dimensionless damping parameter, c^* , which governs the VIV response amplitude.

$$A^* = \frac{A}{D} = \frac{C_L}{\left[\frac{2c_s\omega}{\rho U^2}\right]} = \frac{C_L}{c^*} \quad (3)$$

Where $c^* = \frac{2c_s\omega}{\rho U^2}$.

In equations 1, 2, and 3, c_s is the structural damping constant per unit length; C_L is the average lift coefficient over the entire rigid cylinder in phase with the cross-flow velocity of the cylinder. For the spring-mounted rigid cylinder c^* is the dimensionless parameter which reveals the role that damping plays in regulating response amplitude. This parameter was derived in [Vandiver, 2012] by equating damping forces to lift forces. In that paper it was also shown that equation 3 may be solved for C_L, which provides a simple experimental method for estimating the lift coefficient from free vibration response measurements. This is accomplished experimentally by setting damping and fluid velocity, then measuring A^* , C_L may then be computed, using equation 4.

$$A^*\, c^* = C_L \quad (4)$$

In this paper equilibrium of power is used to derive a parameter similar to c^* , but for the more complex problem of the VIV response of flexible, tension-dominated risers.

POWER FLOW FOR LONG FLEXIBLE RISERS

The equation of motion for a flexible, tension-dominated riser is expressed as,

$$m\frac{\partial^2 y}{\partial t^2} + c\frac{\partial y}{\partial t} - P\frac{\partial^2 y}{\partial x^2} = f(x,t) \quad (5)$$

Where P is the constant tension and $f(x,t)$ is the lift force per unit length. Figure 2 shows that a cylinder is divided into a power-in region with length L_{in} and power out region with length L_{out}. In the power in region, the fluid injects energy into the structure, hence the damping constant c_{in} comes only from the structural damping. In the power out region, the fluid extracts energy, and therefore the damping constant c_{out} is the combination of structural and hydrodynamic damping. The damping ratios in the two regions are defined as,

$$\zeta_{in} = \frac{c_{in}}{2m\omega} \quad (6)$$

$$\zeta_{out} = \frac{c_{out}}{2m\omega} \quad (7)$$

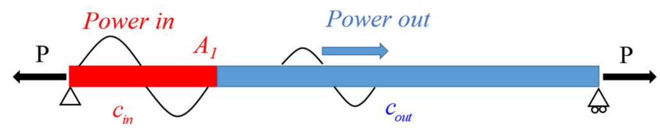

Figure 2. A long top tensioned riser with power-in region at the left end

This paper examines the dynamic behavior assuming VIV excitation in the power-in region. It is assumed that:

1) The energy dissipated by structural damping in the power-in region is much less than the energy lost in the power out region. This is a reasonable assumption for steel risers in common use in the offshore industry. For flexible risers a more complex model may be necessary.

2) The waves leaving the power-in region may be reflected at the far boundary, but die out before returning to the power-in region.

3) Though not essential to the final results, it is assumed for the purpose of ease of computation that the damping is linear and that travelling waves decay exponentially according to the factor $e^{-2\pi\zeta_{out}x/\lambda}$, where λ is the wavelength, ζ_{out} is the damping ratio in the power-out region and x is the distance travelled. The total distance travelled by a wave exiting the power-in region, reflecting from the boundary and returning to the power-in region is $x=2L_{out}$. The ratio of the amplitude of the returning wave to that of the outbound wave is simply given by $A_1/A_2 = e^{-4\pi\zeta_{out}L_{out}/\lambda}$. The ratio of returning wave power to outbound power is in proportion to the square of this ratio. If the returning wave has 10% of the amplitude of the outbound wave, then the returning power is only 1% of the initially radiated power. This is the criterion that is used in this paper to establish when a riser behaves as if of infinite length. If we define n_{out} as the length of the power-out region measured in wavelengths, then $n_{out} = L_{out}/\lambda$. The criterion for infinite length dynamic behavior may be stated as

$$A_1/A_2 = e^{-4\pi\zeta_{out}L_{out}/\lambda} = e^{-4\pi\zeta_{out}n_{out}} < 0.1 \quad (8)$$

This leads to the requirement that

$$\zeta_{out}n_{out} > 0.18 \quad (9)$$

4) The final assumption is that the lift force is a periodic or narrow band random process with the vortex shedding frequency, f_v , determined by the most favorable reduced velocity $U_{r,opt} = \frac{U}{f_v D}$.

Under these assumptions, the balance between time averaged input power-in and output power may be derived.

For a tension-dominated riser, the time averaged output power due to wave radiation can be computed from [Kausel, 2016],

$$\langle \Pi_{out} \rangle = \langle \Pi_{rad} \rangle = \sqrt{Pm}\frac{1}{T}\int_0^T \dot{y}_1{}^2(x,t)dt \quad (10)$$

$\dot{y}_1$ is the cross-flow vibration velocity of the riser at the input end of an infinitely long, tension-dominated cylinder. Since it is assumed that the VIV excitation and response may be characterized as a narrow band random process, then the integral in equation 10 may be expressed as the mean square value of $\dot{y}_1$ [Crandall, 1963] and equation 10 may be written as

$$\langle \Pi_{out} \rangle = \langle \Pi_{out} \rangle_{rad} = \sqrt{Pm} A_{1,rms}^2 \omega^2 \quad (11)$$

If the reader is unfamiliar with this formulation shown in equation 11, another way to arrive at the same conclusion is to compute the radiated wave power as the product of the average energy per unit length and the group velocity. The average energy per unit length for a travelling wave in a tension-dominated beam is the $\langle E \rangle = m A_{1,rms}^2 \omega^2$ and $V_{group} = V_{phase} = \sqrt{P/m}$. The product is exactly the same as shown in equation 11. The theoretical basis for equation 11 is covered in much greater detail in a journal paper which is currently under review.

The input power can be calculated from the integral over the power-in region of the product of the local lift force and the structure's cross-flow velocity.

$$\langle \Pi_{in} \rangle = \frac{1}{T} \oint_{L_{in}} \int_0^T f(x,t) \dot{y}(x,t) dt dx \quad (12)$$

Where $f(x,t) = \frac{1}{2} \rho U^2(x) D C_L(x,t)$.

The actual lift force and hydrodynamic damping processes in the power-in region are complex and not precisely known. The purpose of this paper is not to propose an exact excitation model. The purpose is to show how radiation damping regulates VIV response for high mode number risers. An approximate excitation model, which is consistent with what is generally accepted and is compatible with the formulation used to describe damping is sufficient. It is generally accepted that in the wake synchronized region, the lift coefficient is a narrow band random process with a frequency determined by the most favorable reduced velocity. Therefore, the unknown power computation contained in equation 12 is replaced by the following approximate expression.

$$\langle \Pi_{in} \rangle \approx L_{in} * \frac{1}{2} \rho U_{rms}^2 D C_{L,rms} * A_{rms} \omega \quad (13)$$

The definition of key terms follows. A_{rms} is the spatial and time averaged RMS amplitude in the power-in region.

$$A_{rms} = \sqrt{\frac{1}{L_{in}} \oint_{L_{in}} \frac{1}{T} \int_0^T y^2(x,t) dt dx} \quad (14)$$

For a narrow band random process the RMS velocity is assumed to be approximately equal to $A_{rms}\omega$.

The flow speed in the excitation region is characterized by its mean square value:

$$U_{rms}^2 = \frac{1}{L_{in}} \oint_{L_{in}} U^2(x) dx \quad (15)$$

Within the power-in region, the flow speed is assumed to vary over a narrow range around a mean flow speed. The variation is small, so as to permit wake synchronization at a frequency favored by the local reduced velocity. Such variation in flow speed is typically not greater than plus or minus 15% of the most favorable speed. In this simple model, the lift force in phase with the local cross-flow velocity of the cylinder is characterized by an average RMS lift coefficient within the power-in region. If an exact expression for the lift coefficient were known, the RMS lift coefficient would be computed as follows.

$$C_{L,rms} = \sqrt{\frac{1}{L_{in}} \oint_{L_{in}} \frac{1}{T} \int_0^T C_L^2(x,t) dt dx} \quad (16)$$

Requiring that the input power equal the output power is accomplished by setting equation 11 equal to equation 13.

$$L_{in} \cdot \frac{1}{2} \rho U_{rms}^2 D C_{L,rms} \cdot A_{rms} \omega = \sqrt{Pm} A_{1,rms}^2 \omega^2 \quad (17)$$

Equation 17 is somewhat cumbersome to use, because it involves two different measures of response amplitude. One is the average RMS response in the power-in region and the other is the RMS amplitude of the travelling wave exiting the power-in region. It is assumed here that $A_{rms} \approx A_{1,rms}$. In the numerical simulation this assumption will be shown to be valid. Assuming they are equal, allows equation 17 to be solved for $\frac{A_{rms}}{D}$.

$$\frac{A_{rms}}{D} = C_{L,rms} \frac{L_{in} \rho U_{rms}^2}{2\sqrt{Pm}\omega} = \frac{C_{L,rms}}{c^*} \quad (18)$$

Where $c^* = \frac{2\sqrt{Pm}\omega}{L_{in} \rho U_{rms}^2}$. This is the key result presented in this paper. It says that the average RMS response amplitude in the power-in region is proportional to the lift force, as expected, and inversely proportional to the radiation damping, $\sqrt{Pm}$. Since the radiation damping is proportional to the square root of the tension, the response should decrease as the tension is increased, thus answering the question posed in the title of the paper. When the riser is long enough to behave as an infinite cylinder, then increasing tension will lead to reduced response. This is illustrated in the following numerical simulation.

It was initially assumed that the power lost to structural damping was small compared to the radiation damping. That may be put to a test. The power out due to structural damping in the power-in region is given by

$$\langle \Pi_{out} \rangle_s = \oint_{L_{in}} c_{in} \frac{1}{T} \int_0^T \dot{y}^2(x,t) dt = c_{in} A_{rms}^2 \quad (19)$$

The ratio of this expression to that of the radiated power given in equation 11 is,

$$\frac{\langle \Pi_{out} \rangle_s}{\langle \Pi_{out} \rangle_{rad}} = \frac{c_{in} L_{in} A_{rms}^2}{\sqrt{Pm} A_{1,rms}^2} \approx \frac{c_{in} L_{in}}{\sqrt{Pm}} \quad (20)$$

Where it has again been assumed that $A_{rms} \approx A_{1,rms}$. This expression is evaluated in the numerical example shown below and shown to be much less than 1.0 for typical steel cylinders in sheared flow. The results of this paper concerning the effect of tension on response are insensitive to the amount of structural damping in the power-in region. When the structural damping is not small, it will affect the total response amplitude, but will not change the conclusions regarding the effect of tension on response.

NUMERICAL EXAMPLE

A finite element method is employed to find a numerical solution for equation 1, given a power-in region as shown in Figure 3. A Newmark integration [Bathe, 2006] was used, with the time increment set as Δt=$0.001s$.

- **INFINITE RISER BEHAVIOUR**

Figure 3 shows a long flexible, top tensioned riser in linearly sheared flow. The power in region is centered around the region with higher flow velocity. The properties of the riser model and the flow conditions are listed in Table 1. The physical properties are similar to those of a model that has been tested at Marintek in 2011, and described in [Resvanis et al., 2016] and [Rao et al., 2012]. In this numerical study the damping ratio in the power out region is varied systematically from 1.5% ~ 20% so as to illustrate that the response is insensitive to these values as long as infinite system behavior is observed.

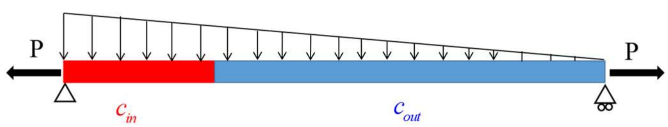

Figure 3. Sketch of a top tensioned riser under sheared flow

Table 1. Properties for riser model in sheared flow

Parameters	Number
Length L (m)	38.00
Power in length L_{in} (m)	7.60
Power out length L_{out} (m)	30.40
Hydrodynamic diameter D (m)	0.012
Cross sectional area A (m^2)	6.91×10^{-5}
Damping ratio in power-in region ζ_{in}	0.3%, 3%
Damping ratio in power-out region ζ_{out}	1.5%, 4%,8%,12%,16%, 20%
Tension P (N)	2000
Stiffness EI (Nm^2)	0
Riser's mass/length (added mass included) m (kg/m)	0.415
Excited mode number n	20
Excited natural frequency ω (rad/s)	115.18
Max. flow velocity U (m/s)	1.4 (linearly sheared flow)
Amplitude of lift coefficient C_{L0}	0.6

It is assumed that the 20$^{\text{th}}$ mode is resonant with the excitation lift force. Hence, the power-in region is 2-wavelengths long, while the power-out region contains 8 wavelengths. The lift coefficient is specified as,

$$C_L(x,t) = C_{L0}\sin(\omega t)\sin\left(\frac{\pi n}{L}x\right) \quad (21)$$

The spatial distribution of lift coefficient is plotted in Figure 4.

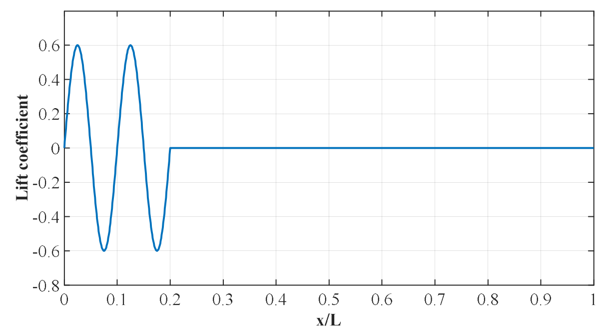

Figure 4. Spatial distribution of input lift coefficient

Figure 5 shows the spatial distribution of non-dimensional vibration amplitude for six values of ζ_{out}. The green dashed line defines the boundary between the power-in and power-out regions. It is shown that when $\zeta_{out}n_{out} > 0.18$, the response in the power-in region is the same for all values of damping in the power-out region. In other words, the response in the power-in region is insensitive to the damping in the power-out region. There is one response curve which is plotted in Figure 5 for which $\zeta_{out}n_{out} < 0.18$. For this case reflected waves of significant amplitude return to the power-in region, which results in standing waves over the entire length of the riser. This case is shown to verify that when the cylinder does not behave as if of infinite length, the response is sensitive to the amount of damping in the power-out region.

Figure 5 is for the case that the structural damping ratio in the power-in region is assumed to like that of steel, 0.3%. The ratio of power lost to structural damping in the power-in region to the radiation damping, $\sqrt{Pm}$, is 0.08. Figure 6 is the same as Figure 5 except that the damping ratio in the power-in region is set at 3%, which is larger than most metal risers. For this case the ratio of the power dissipated due to the structural damping to that due to radiation damping (from equation 20) is approximately 0.76, which is quite large. The large structural damping in Figure 6 results in lower response in the power-in region but does not change the behavior of waves radiated from the power-in region. As long as $\zeta_{out}n_{out} > 0.18$ the response in the power-in region is insensitive to the amount of damping in the power-out region.

It was assumed in the derivations shown for power flow in long flexible cylinders that the ratio $\frac{A_{rms}}{A_{1,rms}}$ would be approximately 1.0. This ratio is plotted in Figure 7 as a function of $\zeta_{out}n_{out}$. The ratio is approximately 0.85 for the cases with light structural damping. For the case of strong structural damping the ratio is approximately 0.95. The value is affected by the lift coefficient model that is used in the simulation, which in this case was a simple standing wave as shown in Figure 4. Rao [2015] has examined experimentally for a model test similar to this simulation and has shown that this ratio is within a few percent of 1.0 for flexible risers with finite length power-in regions.

• EFFECT OF INCREASING TENSION

To illustrate the effect of a change in tension the following conditions are prescribed. The flow velocity is constant, the VIV excitation frequency does not change, but the mode number and wavelength of the resonant mode changes with tension. Figure 8 shows the spatial distribution of the lift coefficient for three different values of tension. The magnitude and frequency of the lift remain the same but the wavelength and resonant mode number change.

Figures 9 and 10 compare the simulated structural response using two values of structural damping ratio (0.3% and 3%) in the power-in region for three values of tension, 2000 N, 2400 N, and 8000 N. The wavelength of the lift coefficient is also varied as shown in Figure 8. This is necessary so that at the three different values of tension the wavelength corresponds correctly to constant value of the VIV excitation frequency. It is shown that the response amplitude decreases as the tension increases. The response amplitude with P=8000 N is approximately half of that with P=2000 N, which agrees quite well with what could be expected from equation 18. It is also shown that the sensitivity to tension is not sensitive to the level of structural damping. It is unlikely that in a real drilling situation the rig could change tension by the amounts in the above simulations. A more realistic scenario is explored in the paragraphs to follow.

Figures 11 and 12 are SHEAR7 predictions of a full scale drilling riser 3000 m in length with buoyancy 1 m in diameter, [Vandiver et al, 2017]. This is a generic test case which was used to compare various VIV response prediction programs in a study which is described in another OMAE2017 paper, [Voie et al, 2017]. The riser properties are given in Table 2. The riser is not tension dominated at the responding mode numbers for the cases considered here. Therefore the radiation damping coefficient is not given simply by $\sqrt{Pm}$ as shown in Equation 11, but is greater due to bending stiffness, [Kausel, 2016]. The power flow due to the radiation of waves is equal to the radiation damping coefficient times the mean square transverse velocity of the riser. The radiation damping coefficient is the real part of the impedance of the riser. A response prediction program, such as SHEAR7, automatically incorporates the effect of bending stiffness. As long as the hydrodynamic and structural damping in the power-out region is great enough to prevent the reflection of significant wave energy back to the power-in region, the results shown in this paper are still valid. The radiation damping coefficient determines the rate of power flow away from the excitation region.

Figures 11 and 12 present the RMS amplitude and fatigue damage rate versus dimensionless length, z/L where z/L=0 is at the bottom of the riser. In Figure 11 the current is a linear shear and in Figure 12 it is a slab flow over the top 20% of the riser. The current profiles are provided in the figures. The predictions have been made with the VIV response prediction program SHEAR7. The minimum tension at the bottom end is 400 kN for the base Case 1. This tension is increased by 20% to 480 kN for Case 2 to show how increasing the tension will affect response amplitude and damage rate in an actual drilling riser practical situation. A 20% increase in tension might be within the capacity of a typical drillship. In all cases shown, the values of $\zeta_{out} n_{out}$ are all greater than 0.18 and therefore, radiation damping controls the damping in the power-out region. No vibration wave energy returns as reflected waves to the power-in region. The damping in the power-out region varies with the local current speed and is computed in the program.

The response is a weighted average response at several modal frequencies using an approach generally referred to as time sharing, when using VIV response prediction programs. The typical power-in region for the sheared flow case in Figure 11 is the top 35% of the riser and the modal frequencies included in the time sharing computation varied from mode 26 to 32. For the slab flow case shown in Figure 12, the power-in region is the top 20% and the responding modal frequencies were from modes 31 to 34. The approximate power-in regions are indicated by a double-headed arrow in each figure. For the sheared flow case, an increase in tension of 20% results in less than 5% decrease in maximum response amplitude and a decrease of approximately 12% in maximum damage rate. For the slab flow case the reductions are slightly smaller. The conclusion drawn is that increasing tension by 20% may lead to small reductions in damage rate and will lead to reductions in response amplitude that will be too small to notice, when observing response in the moon pool.

CONCLUSIONS

1. For long risers that do not see reflected waves return to the power-in region, the wave radiation damping is the dominant damping factor when the structural damping is small.
2. When radiation damping is the controlling damping, increasing tension results in a decrease in the predicted response.
3. On real drilling risers, practical limits on the ability to pull more top tension limit the realistic reductions in damage rate to values that are likely too small to be of much actual significance.
4. For dynamically short drilling risers, in which standing waves along the riser are to be expected, pulling more tension may result in increases or decreases in response, depending on the tuning of resonant modal frequencies to the most favorable reduced velocity.
5. The theoretical basis for this paper is covered in much greater detail in a journal paper which is currently under review.

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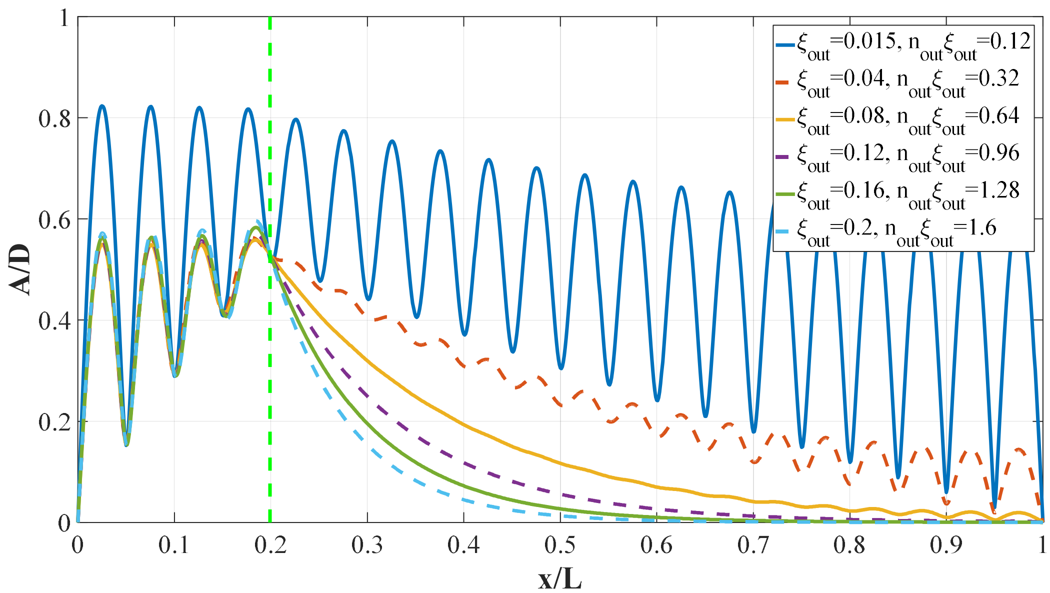

Figure 5. A/D versus x/L for various values of damping in the power out region ($\zeta_{in} = 0.3\%$, *P*=2000 N)

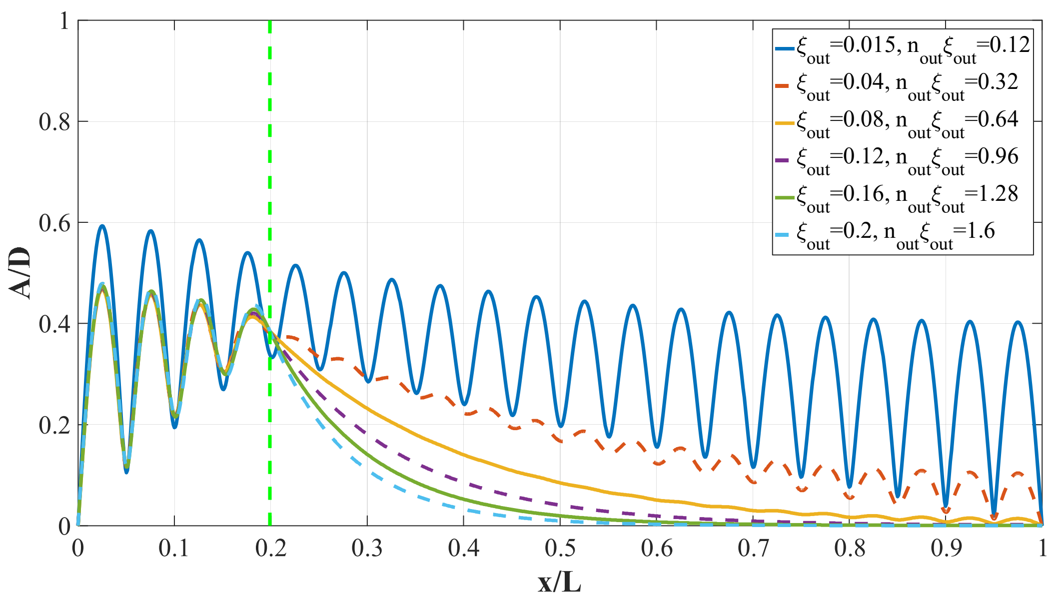

Figure 6. A/D versus x/L for various levels of damping in the power-out region ($\zeta_{in} = 3\%$, *P*=2000 N)

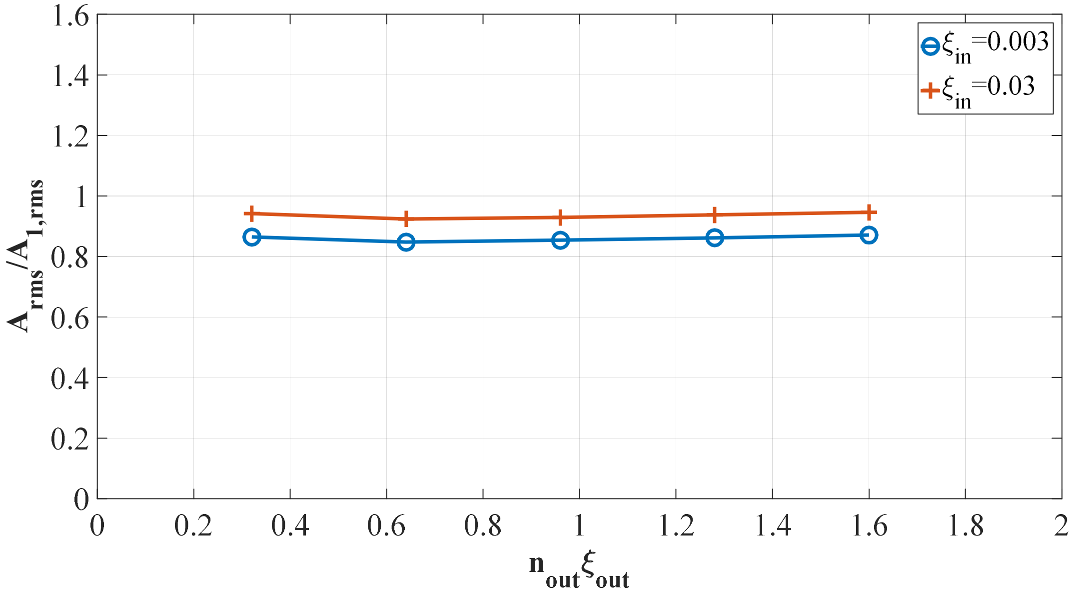

Figure 7. $\frac{A_{rms}}{A_{1,rms}}$ **versus** $\zeta_{out} n_{out}$ **for** ζ_{in}**=0.003 & 0.03**

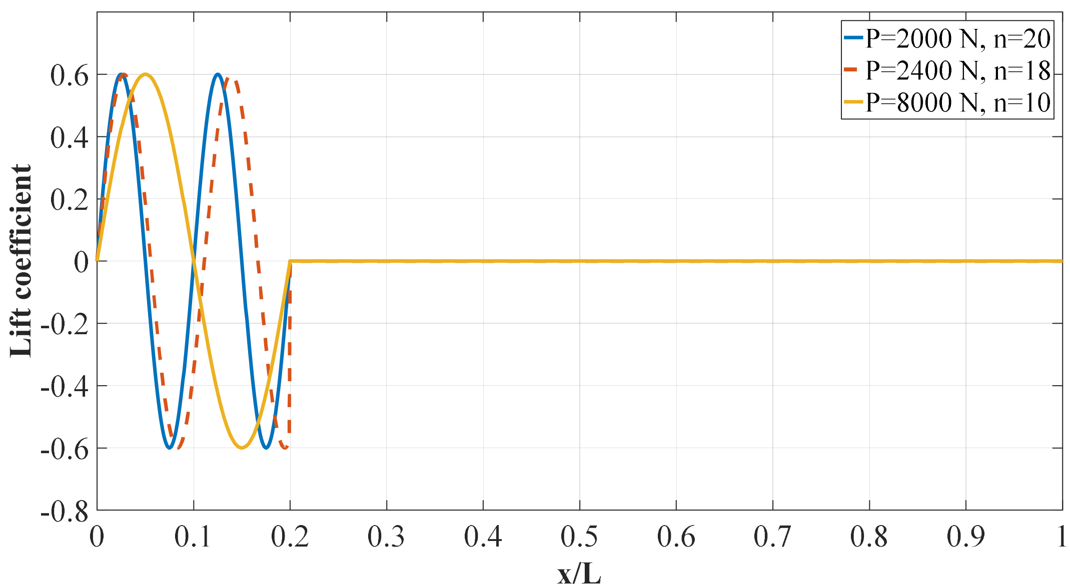

Figure 8. Lift coefficient versus x/L for various tensions

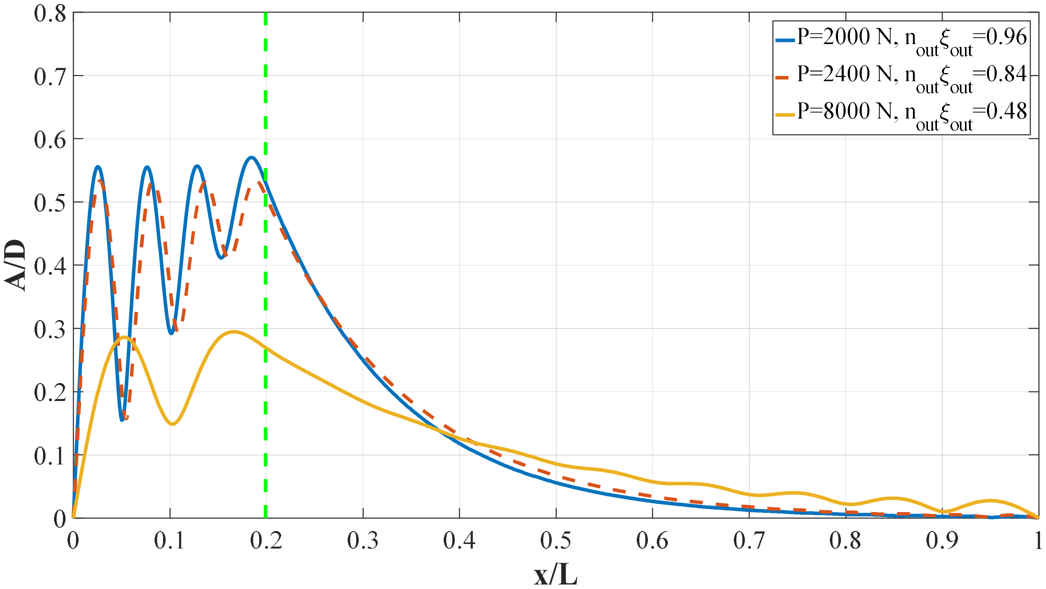

Figure 9. A/D versus x/L for three different tensions ($\zeta_{in} = 0.3\%, \zeta_{out} = 0.12$)

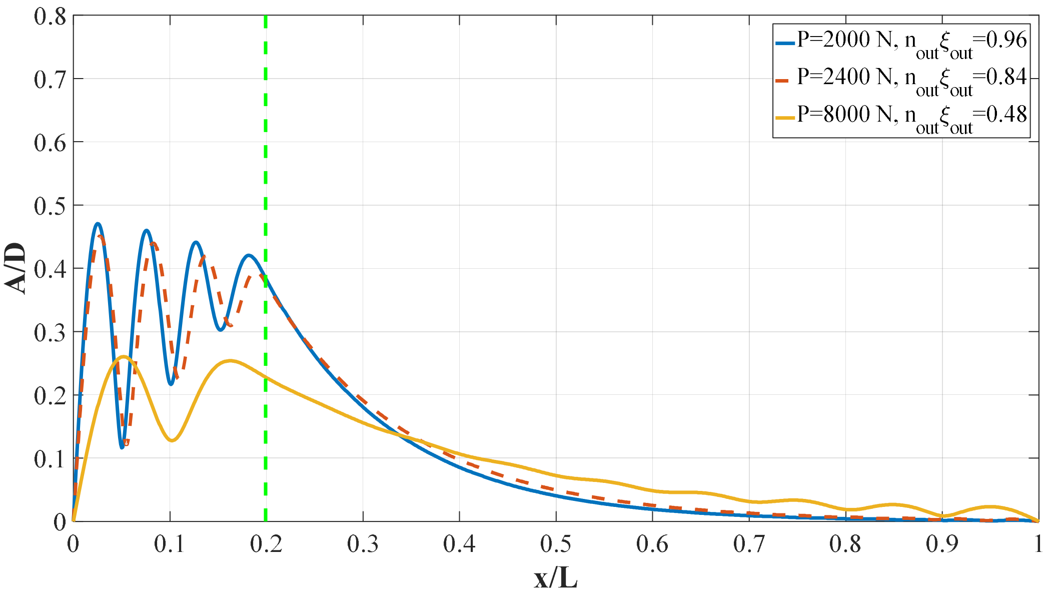

Figure 10. A/D versus x/L for three different tensions ($\zeta_{in} = 3\%, \zeta_{out} = 0.12$)

Table 2. Input parameters for the top tensioned buoyant riser

Parameters	Values
Total length between pinned ends L (m)	3012
Outer diameter in bare regions (m), 70 m at the top, 70 m at the bottom	0.6
Outer diameter in buoyant regions (m)	1
Tension at the lower end P (kN) Case 1, Case 2	400, 480
Tension at the upper end (kN), Case 1, Case 2	1699, 1779
Stiffness EI (kNm^2)	316051
Structural damping ratio in power-in and power-out regions, ζ	0.3%
m, Total dynamic mass/length in bare regions (kg/m), includes added mass with $C_a = 1$	920
m, Total dynamic mass/length in buoyant region (kg/m), includes added mass with $C_a = 1$	1640
SHEAR7 input parameters S_t, dV_r, C_L table, cutoff	0.18, 0.4, 2, 0.5

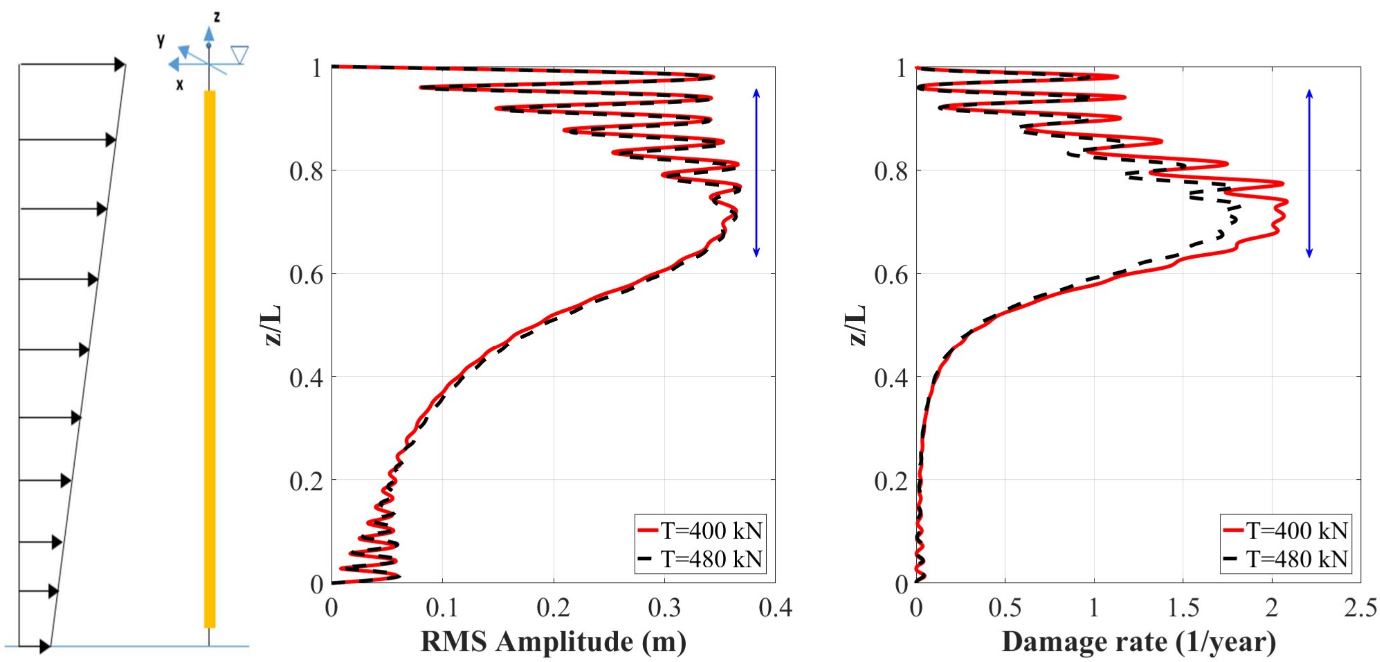

Figure 11. A/D versus z/L for two different tensions under linearly sheared flow ($U_{max} = 0.9\text{m/s}$, $U_{min} = 0.1\text{m/s}$)

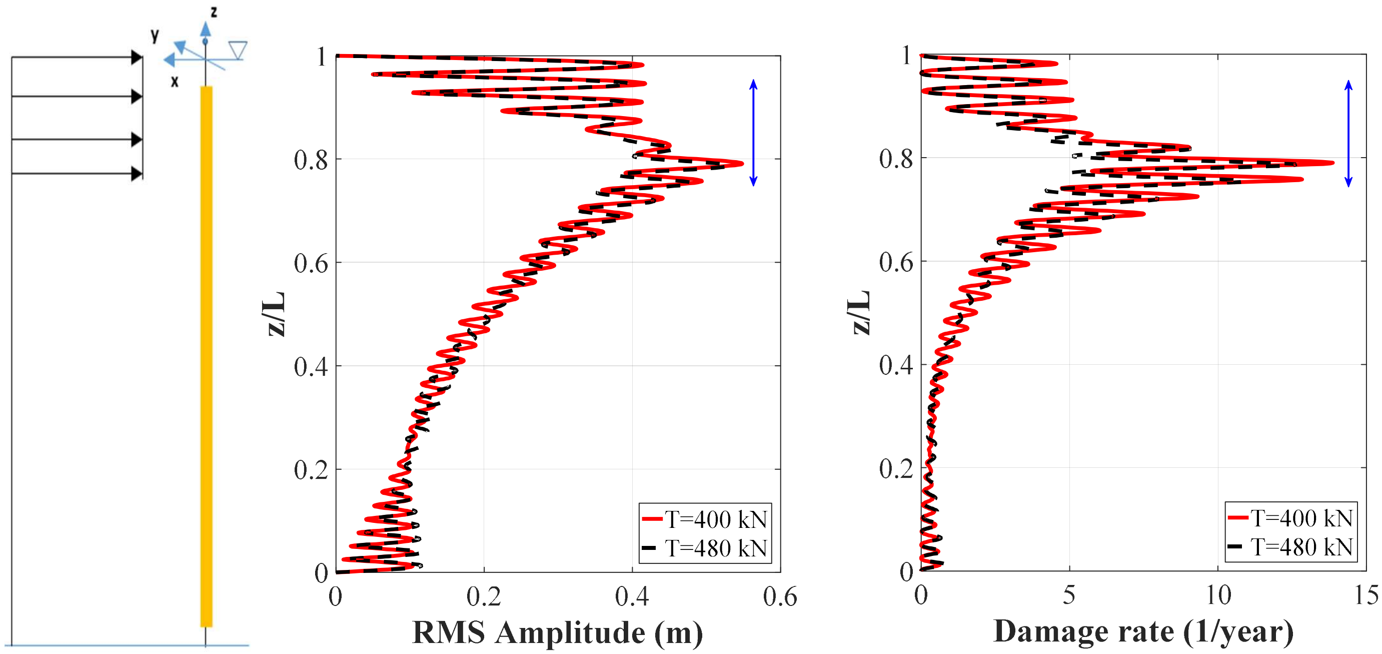

Figure 12. A/D versus z/L for two different tensions under slab flow ($U = 0.9$ m/s at top 20%)